



Integration of a wood pellet burner and a Stirling engine to produce residential heat and power



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HIGHLIGHTS

- The position of the Stirling engine in relation to the burner is highly important.
- The burner cycling operation affects the Stirling engine hot side temperature.
- The thermal power absorbed by the engine increases with higher hot side temperature.
- Start-up and steady-state losses and power output are presented.
- The overall system efficiency reached 72%.

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ABSTRACT

The integration a Stirling engine with a pellet burner is a promising alternative to produce heat and power for residential use. In this context, this study is focused on the experimental evaluation of the integration of a 20 kW_{th} wood pellet burner and a 1 kW_e Stirling engine. The thermal power not absorbed by the engine is used to produce hot water. The evaluation highlights the effects of pellet type, combustion chamber length and cycling operation on the Stirling engine temperatures and thermal power absorbed. The results show that the position of the Stirling engine is highly relevant in order to utilize as much as possible of the radiative heat from the burner. Within this study, only a 5 cm distance change between the Stirling engine and the pellet burner could result in an increase of almost 100 °C in the hot side of the engine. However, at a larger distance, the temperature of the hot side is almost unchanged suggesting dominating convective heat transfer from the hot flue gas. Ash accumulation decreases the temperature of the hot side of the engine after some cycles of operation when a commercial pellet burner is integrated. The temperature ratio, which is the relation between the minimum and maximum temperatures of the engine, decreases when using Ø8 mm wood pellets in comparison to Ø6 mm pellets due to higher measured temperatures on the hot side of the engine. Therefore, the amount of heat supplied to the engine is increased for Ø8 mm wood pellets. The effectiveness of the engine regenerator is increased at higher pressures. The relation between temperature of the hot side end and thermal power absorbed by the Stirling engine is nearly linear between 500 °C and 660 °C. Higher pressure inside the Stirling engine has a positive effect on the thermal power output. Both the chemical and thermal losses increase somewhat when integrating a Stirling engine in comparison to a stand-alone boiler for only heat production. The overall efficiency of the pellets fired Stirling engine system reached 72%.

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1. Introduction

An increased use of biomass as a renewable fuel contributes to keep down the greenhouse gas emissions especially if modern biomass conversion technologies are employed [1]. Around 2.6

billion (short scale) people depend on the traditional use of biomass for cooking and heating; these people represent around 38% of the world population [2]. The traditional use of biomass generates high emission levels of particles and carcinogenic compounds, both affecting the health negatively [3]. The situation for these people would improve with the access to modern biomass conversion technologies with high efficiency and low emission levels [1,2]. Biomass based technologies for residential use such as wood pellets stoves and boilers have reached the European market successfully [4]. The widespread use of these units would be more beneficial if they produce not only heat but also electricity. According to IEA [2], around 20% of the world's population had no access to electricity in 2009. These people are mostly located in rural areas of developing countries and are thus an unexplored market in need of better energy solutions to satisfy their energy demands.

In this context, the combined production of useful heat and electricity (CHP) based on the integration of a pellet combustion technology and a prime mover could be a good alternative [5]. Among prime mover technologies, the Stirling engine is an attractive option in the CHP system [6], especially in micro-CHP plants, since Stirling engines can be manufactured in a low power range of 1–10 kW_e [7] suitable for residential use. The Stirling engine has several advantages; for example high thermal efficiency [8], low noise and low emission levels [9]. Most of all, it is fuel flexible, i.e. whatever heat source can be used such as solar energy, biomass, biogas or fossil fuels [6]. However, several drawbacks need to be overcome, among them: the high investment cost [9], the low commercial availability [10] and some technical problems due to leakage of the working fluid [10], issues related to the testing of new biomass fuels (fouling and maintenance) [5] and the use of high temperature resistant materials [10]. In comparison to a diesel or gas engine, the Stirling engine is, with today's drawbacks, more dependent on regular service and maintenance and will therefore have somewhat lower accessibility, but has the large advantage of being able to run directly on biomass fuels.

The Stirling engine is an external combustion engine with a pressurized working fluid inside. This fluid expands when it is heated and drives a piston producing mechanical work [11]. Depending on the placement of the shafts and piston, Stirling engines are divided into alpha, beta and gamma types [6]. Several parameters affect the performance of the Stirling engine. Among them: the mechanical configuration of the engine, the thermal and physical properties of the working fluid, the temperature difference between the cold and hot sides, the pressure of the engine and the effectiveness of the regenerator, heater and cooler [10]. Several authors studied experimental parameters like temperatures of the hot source or the hot side of the engine, speed, torque, pressure, power output and efficiency. Karabulut et al. [10] showed experimental values of power output, torque and efficiencies as a function of variables such as speed and hot end temperature in the Stirling engine. Cinar et al. [11] showed the same experimental values but as a function of the hot source temperature. They found that with an increase in the hot end and the hot source temperatures, higher power output is achieved. Numerical studies have also been performed to optimize the Stirling engine design. Thombare and Kamare [12] performed simulations and experiments and found that a higher amount of heat supplied to the engine results in lower temperature ratios (relation between the minimum and maximum temperatures of the Stirling engine). They also found that higher thermal efficiency is obtained at higher effectiveness of the regenerator. Timoumi et al. [13] developed a dynamic model (including several sources of losses) of a prototype rhombic Stirling engine which has led to a reduction

of losses and improvement of the engine performance. Parlak et al. [14] executed a thermodynamic model to maximize the power output and thermal efficiency of a gamma-type Stirling engine. The results of that model in terms of power output and efficiency were not validated against experimental data. Based on this review, further analysis of the effect of operating variables on the performance of the Stirling engine is of high importance especially to develop more accurate numerical models and to further develop the Stirling engine technology.

The fuel flexibility of the Stirling engine depends on the type of heat source technology adapted. Some CHP systems based on Stirling engines are available in the market but they are mainly fueled on gaseous fuels; i.e. they are adapted to gas burners such as the unit commercialized by WhisperGen Company [15]. This unit has been compared with other micro CHP units based on internal combustion engines and a conventional natural gas fired boiler by Rosato et al. [16]. They have concluded that this unit and the others allow for reducing primary energy use, carbon dioxide equivalent emission and operating costs compared to the conventional natural gas fired boiler but also for a given tank volume the unit does not give the best performance. Benchmark tests were performed to evaluate two micro-CHP units based on Stirling engines [17]. One unit that is fueled by natural gas and the other fueled by biogas were considered in these tests. Only the first one, the SOLO Stirling Micro-CHP unit, met the “Der Blaue Engel”, which is the German environmental label regulation [18] and presented high overall efficiencies and low emission levels. The second one, a SM5A micro-CHP manufactured by Stirling Denmark, presented low efficiencies. In another recent study, some improvements in a biogas-fed Stirling co-generator have been analyzed by proposing the addition of a spiral gas–gas heat exchanger raising the electrical efficiency to up to 22.5% [19]. Currently, Stirling DK [20] commercializes a 35 kW_e Stirling engine either as stand-alone or as a part of a CHP unit fueled on gasified biomass. Genostirling [21] manufactures 1 to 3 kW_e Stirling engines of alpha and gamma types which can be adapted to different heat sources as gas burners and as proposed in this study with a pellet burner. The investment cost of this engine is around 11,000 Euros per kW_e (for a 1 kW_e engine), which is much higher than many other power generation technologies. A larger engine of 3–3.5 kW_e could cost 3700–6400 Euros per kW_e depending on whether it is operating on solar irradiation or fuel combustion [22]. Microgen Engine Corporation [23] produces a 1 kW_e free piston Stirling engine which is being used with natural gas but is presently investigated to use other heat sources such as biomass. Further testing of the reliability and availability of Stirling engines is required when using solid biomass as fuel.

Some numerical and experimental studies have focused on the use of the Stirling engine fueled on wood in the form of pellets or powder. Kuosa et al. [24] introduced a fouling factor to numerically evaluate the performance of an alpha type Stirling engine. That evaluation was performed in terms of the power output, the brake efficiency and the recovery of heat as a function of the fouling factor. Nishiyama et al. [25] performed combustion tests to monitor the quality and efficiency of the wood powder combustion before the integration of a 55 kW_e Stirling engine. They found that at air fuel ratios over 2, the Stirling engine can produce theoretically 55 kW_e. They also highlighted the relevance of the air–fuel ratio to maximize the power output of the engine due to the relevance of the hot end temperature on the performance of the engine. In another study performed by Sato et al. [26], the same 55 kW_e Stirling engine as above, was tested in a CHP unit fueled on wood powder. The tests revealed ash fouling problems in the heat exchanger of the engine. The authors suggested therefore to optimize the combustion temperature and the inlet gas

temperature of the engine as well as to develop a hot ash cleaning system. However, the problem with a filter in between the burner and the Stirling engine is that the temperature on the hot side and thereby the power output will be significantly affected. Thiers et al. [5] performed experimental tests using a commercial micro-CHP unit based on an alpha type Stirling engine of 1.5–3 kW_e fueled on wood pellets. The engine was manufactured by Sunmachine GmbH. The aim of that study was to develop a numerical model under transient conditions. They used the data of a test bench to characterize the annual performance if this system could be implemented in a building. The CHP unit presented low efficiencies and large heat losses since the manufacturer specifications were not reached.

Direct combustion coupled with a Stirling engine could be a more simple technology than a gasifier and an internal combustion engine and therefore, could be more suitable for isolated regions [25]. In comparison with a gasifier system, less need of man power due to the automatic feeding and control and less risk of CO poisoning of workers are among the advantages of using a pellet burner and a Stirling engine [27]; however, ash accumulation need to be considered and further related with the performance.

Based on this review, there is a need of further studies of the performance of CHP systems with Stirling engines fueled on wood pellets and other biomass fuels to approach commercialization and a more wide-spread use of this technology, thus reducing the investment cost. One important parameter to study is the hot side temperature of the engine as a function of combustion temperature and cycling operation of a commercial pellet burner; a relation not covered in deep in previous studies. The temperature ratio and effectiveness factor of the regenerator as a measure of the efficiency of heat transfer inside the engine is also of importance and previously not related to the hot side temperature and pressure. By quantifying the losses in a heat and power system empirical relations for modeling under steady-state and transient conditions can be found.

This study is mainly focused on the experimental evaluation of the integration of an overfed wood pellet burner [28] and a prototype gamma Stirling engine [21], in which several new parameters are considered to reflect some of the real operation conditions of a CHP plant. The type of heat source has a great impact on the Stirling engine hot side temperature which in turns may affect the heat transfer process and therefore should be evaluated. This study identifies the influence of the pellet size and the effect of the distance between the burner and hot side of the Stirling engine, which highlights the importance of where to place the engine in relation to the pellet burner. This has not been examined by other studies. The impact of the cycling operation of the burner on the hot side temperature under transient conditions is also presented. This is specifically important to take into consideration, since the cycling behavior reflects the real operation condition at longer runtime of a pellet-fired Stirling engine. The heat transfer efficiency of the Stirling engine is evaluated in terms of the temperature ratio and effectiveness of the regenerator. The effect of the hot side temperature on the thermal power absorbed by the engine is also assessed for different operating pressures. Finally, the losses and power outputs of the system are presented under transient conditions. The overall efficiency of the system with and without the Stirling engine (previous results [29]) is compared under steady-state conditions. CO and NO_x are also considered and compared with previous studies of the system without the Stirling engine [29]. The evaluation of the Stirling engine and the overall system within this paper provides new valuable data for modeling studies and further development of the Stirling engine technology together with a pellet burner.

2. Methodology

2.1. Fuel analysis

Wood pellets of Ø6 mm and Ø8 mm were used in the experiments [30]. Table 1 shows the chemical composition, heating values and physical characteristics of the pellets used in the experiments [27]. The chemical composition and heating values were determined by an accredited laboratory: Bränslelaboratoriet Umeå AB, in Sweden [31]. This laboratory has provided the relative measurement uncertainties. The expanded uncertainties presented in Table 1 were calculated with a coverage factor of 2. Physical characteristics of individual pellets such as the length, diameter, width, height, mass and particle density were determined by taking the average data of randomly selected 100 pellets [27]. The average bulk density was obtained by measuring the mass of three samples with a normal laboratory balance and the volume using a graduated cylinder [27]. The water content was measured according to ÖNORM M 7135 [32] before the experiments were performed for this study.

2.2. Description of the experimental set-up and procedure

The integration of the main parts of the experimental set-up is shown in Fig. 1a. A commercial wood pellet burner of 20 kW_{th} nominal thermal capacity [28] was integrated with a prototype 1 kW_e Stirling engine [21] and a residential boiler of the same thermal capacity of the burner. The Stirling engine is a two cylinder gamma type that has been previously tested with LPG by the manufacturer. The Stirling engine is equipped with a hot heat exchanger which is exposed to the hot flue gas inside an insulated box shown in Fig. 1a. In this figure, the distance between the hot side heat exchanger of the Stirling engine and the pellet burner (D) is indicated. The Stirling engine also has a cold heat exchanger (the heat sink) where cold water passes through at a rate of 1.5 L/min. The engine can work with different working fluids such as air, nitrogen and helium. In this study nitrogen was used, since tests with air showed higher accumulation rate of oxides inside the engine. The maximum pressure allowed inside the engine is 25 bar. The maximum temperature that the hot side of the Stirling engine can resist is 1000 °C since the material of the hot side is AISI 310. The Stirling engine is also connected to a DC permanent magnet

Table 1

Chemical composition, heating values and physical characteristics of the wood pellets used in the experiments with the uncertainty inserted [27].

Chemical composition	Wood	
Carbon (wt% d.b.)	50.4 ± 3.0	
Hydrogen (wt% d.b.)	5.9 ± 0.5	
Nitrogen (wt% d.b.)	<0.1 ± 0.03	
Oxygen (wt% d.b.)	43.3	
Ash (wt% d.b.)	0.3 ± 0.1	
Sulfur (wt% d.b.)	<0.010 ± 0.002	
Heating values		
HHV (MJ/kg d.b.)	20.3 ± 0.4	
LHV (MJ/kg d.b.)	18.9 ± 0.4	
Physical characteristics	Wood pellets (Ø8 mm)	Wood pellets (Ø6 mm)
Diameter (mm)	8.2 ± 0.2	6.3 ± 0.2
Length (mm)	11.2 ± 2.7	12.8 ± 4.3
Mass/pellet (g)	0.6 ± 0.3	0.4 ± 0.2
Moisture content (%) ^a	6.8 ± 0.05	7.0 ± 0.05
Particle Density (kg/m ³)	1043 ± 454	1020 ± 185
Bulk density (kg/m ³)	Ca 637	Ca 604

^a This study.

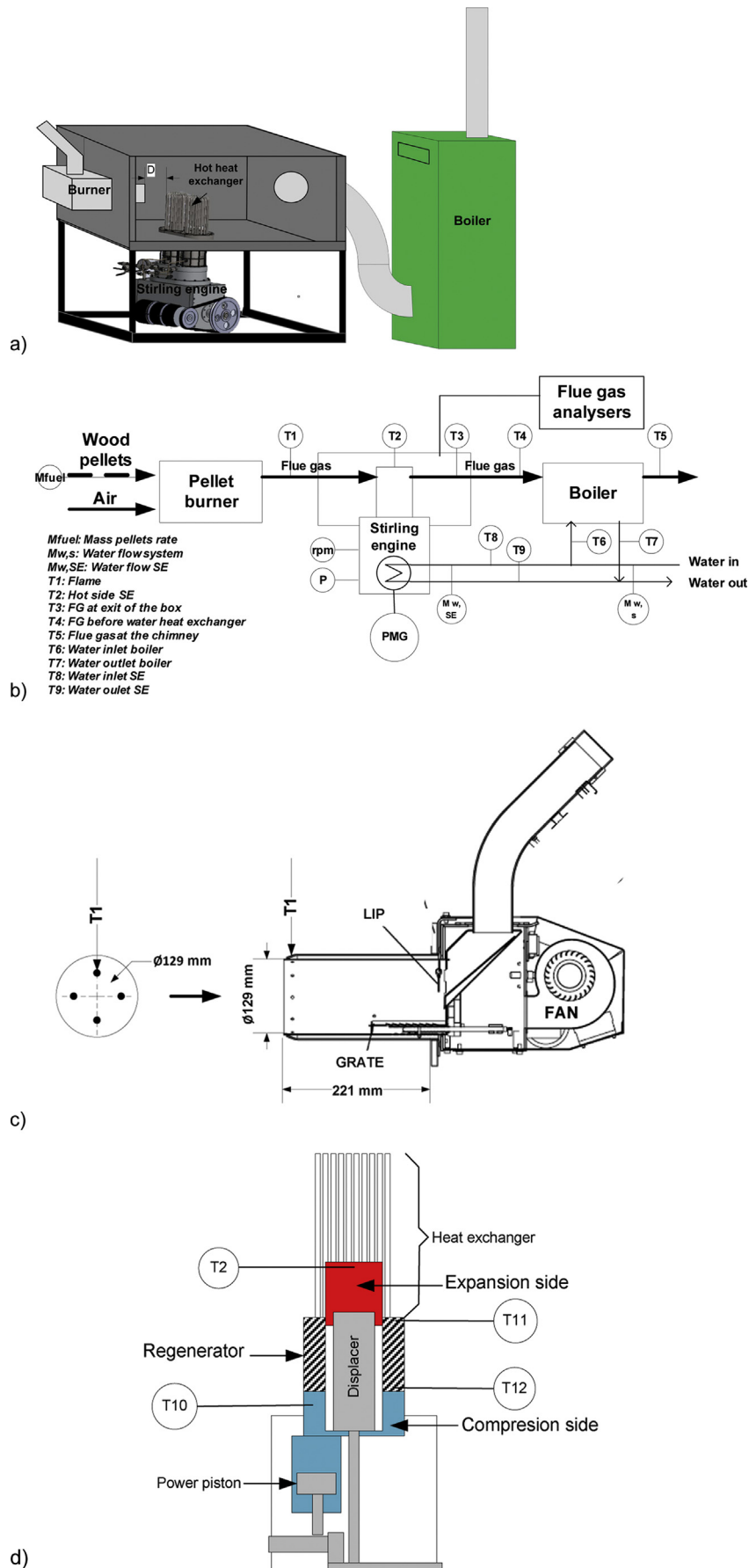


Fig. 1. a) Experimental set-up, b) measurement points, c) pellet burner dimensions and positions to measure T_1 , d) scheme of Stirling engine temperatures measurement points.

generator (PMG) through a flywheel. The residential boiler produces hot water where the temperature of the water must not exceed 90 °C. The water of the boiler and Stirling engine is cooled down in a plate heat exchanger by water from the Stockholm district cooling network.

Fig. 1b shows a block diagram with the main measurement points. The path of the flue gas is as follows: the flue gas goes through the hot side of the Stirling engine and then is carried to a pipe. After this pipe, the flue gas enters the boiler where the water absorbs a large part of the heat from the flue gas. After that, the flue gas is evacuated to the lab hall exhaust system at temperatures between 100 °C and 180 °C. The exhaust system is controlled by a fan regulated from 10 to 100% of fan power. The power of this fan was kept constant during every test.

Temperature T_1 (Fig. 1b) corresponds to the flame temperature. A type K, Ø1.5 mm, Inconel 600 thermocouple was used to measure T_1 . This thermocouple was moved and placed according to the arrangement indicated in Fig. 1c in the cross section of the inner burner tube in front of the grate. Average flame temperature was considered by the four temperatures measured. Fig. 1d shows the location some thermocouples inside the Stirling engine: T_2 is the temperature of the hot side end; T_{10} is the temperature of the hot side of the regenerator; T_{11} is the temperature of the cold side of the regenerator and T_{12} is the temperature of the cold side end. The temperatures shown in Fig. 1b and d were measured using type K thermocouples of Ø1 mm. According to the type of thermocouple, the expanded uncertainty is ± 3.2 °C with a coverage factor of 2.

The water flow of the boiler system was monitored using a Coriolis flow meter ($\pm 0.1\%$). The water flow to the Stirling engine was measured using a rotameter ($\pm 1\%$).

The flue gas compounds were continuously monitored using a non-dispersive infrared analyzer for CO and CO₂ (URAS 10P), a magnetic pressure analyzer for O₂ (Magnos 6G), and a chemiluminescent analyzer for NO_x (CLD 700 EL ht). Before each test, the instruments were calibrated with zero and span gases with an uncertainty of $\pm 2\%$ vol.

A pulse sensor and a frequency to analog converter (OMRON E2A, and Red Lion IFMA) were used to monitor the speed of the crankshaft of the Stirling engine with an uncertainty of $\pm 0.2\%$. A pressure transducer (RS type 46) with an analog signal was used to measure the pressure inside the engine with an uncertainty of ± 0.1 bar.

Temperatures, concentration of flue gas compounds, pressure and speed of the engine were recorded every three seconds using a data logger from the beginning to the end of each test.

The Stirling engine ran more than 100 h (3–9 h per day) at low pressure (8–12 bar) following the instructions of the manufacturer. During this period of time, the optimum distance from the burner to the Stirling engine was found by performing three experiments per distance test. After 100 h of operation, the pressure of the engine was increased between 12 and 18 bar. No load was applied to the engine in order to study the heat transfer effects. The cycle of operation of the pellet burner was of 3 h before the

ash removal phase. Average data was calculated under steady-state conditions and standard deviations were estimated based on Taylor and Kyatt [33]. The intervals of the tests are explained in each section below.

Table 2 shows the results of tests performed by the manufacturer of a similar Stirling engine [21] at 18 bar (air). They used a LPG burner with a thermal capacity of 7.5 kW as a heat source. The thermal power output of the Stirling engine is considered as the heat recovered by the water sink. The relation between the electrical and thermal power is the electricity to heat ratio measured and corresponds to 0.3 for this engine. This relation is assumed to be constant at any pressure within this study to estimate the electricity production.

2.3. Analysis of the system

Fig. 2 illustrates the energy flows in the overall system and the subsystems 1 (pellet burner), 2 (Stirling engine) and 3 (boiler). The input power to the system corresponds to the fuel power input (P_{Fuel}). The useful power output of the system is the thermal power absorbed by the water boiler and the overall power output of the Stirling engine in form of thermal and electrical power.

The fuel power input is estimated based on the LHV of the fuel and the consumption ($\dot{m}_{\text{Fuel}}$) on dry basis according to Eq. (1).

$$P_{\text{Fuel}} = \text{LHV} \cdot \dot{m}_{\text{Fuel}} \quad (1)$$

2.3.1. Overall efficiency

The overall efficiency of the system is evaluated as the ratio between the heat recovered by different means (as hot water and thermal and electrical power of the engine) and the fuel input power using Eq. (2).

$$\eta_{\text{system-with-SE}} = \frac{P_{\text{w,boiler}} + P_{\text{SE}}}{P_{\text{fuel}}} \cdot 100 \quad (2)$$

where: the thermal power absorbed by the water of the boiler ($P_{\text{w,boiler}}$) is found by:

$$P_{\text{w,boiler}} = \dot{m}_{\text{w,boiler}} \cdot C_{p,\text{w,boiler}} \cdot (T_7 - T_6) \quad (3)$$

The overall power output of the Stirling engine (P_{SE}) is considered in Eq. (4) as the thermal power absorbed by the water calculated according to Eq. (5) and 30% of this value as declared by the manufacturer of the Stirling engine [21] as electrical power.

$$P_{\text{SE}} = P_{\text{w,SE}} + 0.3 \cdot P_{\text{w,SE}} \quad (4)$$

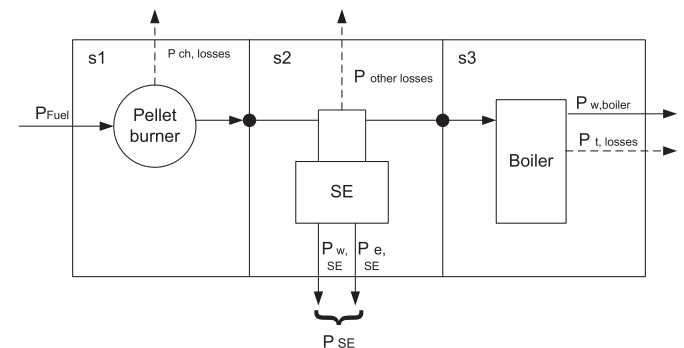


Fig. 2. Schematic of energy flows in the system.

Table 2
Test performed by the manufacturer of the Stirling engine at 18 bar (air) [21].

Parameters	
Test fuel	LPG
Fuel power input	7.5 kW
Thermal power output of the boiler	2.8 kW
Thermal power output of the Stirling engine	2.3 kW
Electrical power of the Stirling engine	0.7 kW
Thermal losses in the chimney	1.2 kW
Electricity to heat ratio	0.3

$$P_{w,SE} = \dot{m}_{w,SE} \cdot C_{p,w,SE} \cdot (T_9 - T_8) \quad (5)$$

2.3.2. Quantified losses in the system

The chemical losses (P_{ch}) due to CO emission and thermal losses due to hot flue gas at the chimney (P_t) are calculated according to SS-EN 14785:2006 [34] as:

$$P_{ch} = \frac{\dot{m}_{fuel} \cdot 12644 \cdot CO \cdot (C_{fuel} - C_{ash})}{[0.536 \cdot (CO_2 + CO) \cdot 100]} \quad (6)$$

$$P_t = \dot{m}_{fuel} \cdot (T_5 - T_{room}) \cdot \left[C_{p,md} \cdot \frac{(C_{fuel} - C_{ash})}{(0.536) \cdot (CO_2 + CO)} + \left[C_{p,mH_2O} \cdot 1.92 \cdot \frac{(9H + M)}{100} \right] \right] \quad (7)$$

The mass flow of the flue gas is calculated according to Eq. (8) [34].

$$\dot{m}_{FG} = \frac{\dot{m}_{fuel} \cdot 1.3 \cdot (C_{fuel} - C_{ash})}{0.536(CO_2 + CO)} + \frac{9H + M}{100} \quad (8)$$

2.3.3. Temperature ratio of the Stirling engine

The temperature ratio of the Stirling engine is defined as the ratio between the minimum and maximum absolute temperatures of the fluid inside the engine [12], which according to the nomenclature in this study is (temperature in K):

$$R = \frac{T_{10}}{T_2} \quad (9)$$

2.3.4. Effectiveness of Stirling engine regenerator

The effectiveness of the regenerator is calculated using Eq. (10). This equation considers the ratio of the difference between the temperatures measured in inlet (T_{11}) and outlet of the regenerator (T_{12}); and the difference between the maximum (T_2) and minimum temperatures (T_{10}) [12].

$$Ef = \frac{T_{11} - T_{12}}{T_2 - T_{10}} \quad (10)$$

3. Results and discussion

3.1. Influence of pellets type, distance between the burner and hot side of the Stirling engine and cycling operation

Fig. 3a displays the relation of the distance between the burner and the Stirling engine hot side (D) and the temperature in the hot side of the engine. Ø8 mm wood pellets were used for these tests with a maximum difference in the fuel consumption of $\pm 5\%$ and an average flame temperature of 1013°C ($\pm 78^\circ\text{C}$). It is observed that the temperature in the hot side of the Stirling engine is very sensitive to the distance from the burner since only 5 cm of increased distance could give rise to almost 100°C temperature reduction at the hot side. However, the distance becomes of less importance when increasing it further, as shown in Fig. 3a. It seems that at closer distance (32 cm) the radiative heat transfer plays an important role, while at further distance increase, convective heat transfer plays a larger role. Higher power output is expected at

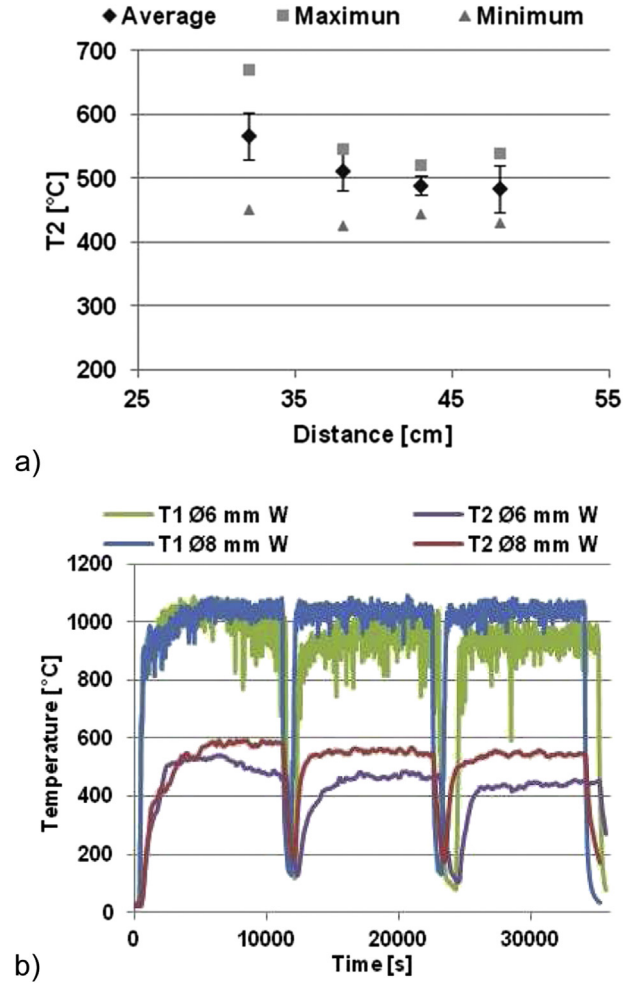


Fig. 3. a) Temperature in the SE hot side (T_2) as function of the distance between the burner and the SE (with standard deviation bars), b) Flame (T_1) and hot side of the SE temperatures (T_2) over 9 h test using Ø6 mm and Ø8 mm wood pellets, 12 bar, 37 cm distance.

higher temperatures at the hot side of the Stirling engine [10], and when operating with a pellet burner, the radiation of the flame improves this temperature. However, a shorter distance of the engine to the burner may in the long run affect the heat transfer negatively due to the accumulation of ash on the tubes of hot side heat exchanger. The sensitivity to the distance shows that if integrating a hot particulate filter in between the burner and the engine, the temperature in the hot side heat exchanger will clearly be affected.

The effect of ash accumulation on the hot side temperature (T_2) is shown in Fig. 3b where, after 9 h of continuous operation, this temperature has decreased. Small-scale burners have a cycle of operation in which automatically ash removal phases are included in time intervals. For overfired burners of 20 kW_{th} , the ash is automatically pushed away from the grate with air after three hours of operation as seen in Fig. 3b. The inorganic compounds of the fuel may also cause fouling on the surface of the heat exchanger, reducing the heat transfer from the hot flue gas to the fluid inside the engine [26]. Among the inorganic compounds of the fuel, potassium (generally included in wood ash) is released in the high temperature burning zone as vapor and may condense on the heat exchanger tube surface due to low temperatures ($<1000^\circ\text{C}$). KOH and K_2SO_4 can be also formed if sufficient sulfur is available [35].

Table 3

Data to assess (input) steady-state flue gas flow rate and temperatures of the flame and of the hot side end SE with standard deviation inserted.

	Fuel mass flow [kg/h]	C_{fuel} [wt%]	C_{ash} [wt%]	H [wt%]	M [wt%]	CO_2 [vv%]	CO [vv%]	F.G. mass flow [kg/h]	T_1 [°C]	T_2 [°C]
Ø8 mm	4.3 ± 0.01	46.9 ± 2.8	0.7 ± 0.1	5.5 ± 0.5	6.8 ± 0.05	8.3 ± 0.8	0.01 ± 0.003	59.4 ± 6	1036 ± 19	557 ± 11
Ø6 mm	3.8 ± 0.01	46.9 ± 2.8	0.4 ± 0.1	5.4 ± 0.8	7.0 ± 0.05	7.1 ± 0.87	0.01 ± 0.01	60.2 ± 8	968 ± 39	467 ± 19

Both KOH and K_2SO_4 may form deposits (fouling) on the surface of the heat exchanger. Therefore, this type of fouling is likely to occur in the Stirling engine heat exchanger due to the temperature measured on hot side is below 1000 °C; but this type of fouling will be notable after months of continuous operation. There is trade-off between the distance and fouling issues regarding the expected power output of the Stirling engine which requires to be considered to select the configuration of the system.

In Fig. 3b, the difference in temperatures T_1 and T_2 as function of pellets type is also seen. Higher temperatures are measured when using Ø8 mm wood pellets which are due to the higher density of these pellets. This means a larger mass of pellets burning onto the grate for the same volumetric feed. The results in Table 3 show a slightly higher amount of gases resulting from the combustion of Ø6 mm wood pellets (higher air flow). This is mainly due to a higher burn out explained by the lower percentage of unburned carbon for the Ø6 mm wood pellets (Table 3). Previous results [29] indicated that the ignition of the Ø6 mm wood pellets is faster and therefore, there is a rapid consumption of fuel which may locate the flame further away from the hot side of the Stirling engine. However, the flame length was the same for both pellets types as observed during the experiments of this study. It is expected that when using Ø8 mm wood pellets, higher power output of the Stirling engine is achieved.

3.2. Thermal power analysis of the Stirling engine

The temperature ratio and the effectiveness of the regenerator as a function of pellets type, cycling operation of the burner and operation time of the engine are shown in Table 4. The ratio and effectiveness parameters affect directly the thermal efficiency of the engine as found in some numerical [36] and experimental studies [12]. These studies have indicated that to maximize the thermal efficiency and power output of the engine there should be a higher amount of heat supply to the engine, which is notable at lower temperature ratios [36]. In Table 4, it is observed that the temperature ratio is higher for the combustion of the Ø6 mm wood pellets in comparison to the Ø8 mm pellets. This means that the difference between the maximum and minimum temperatures is lower for the Ø6 mm wood pellets and therefore, there is a lower amount of heat supply to the engine which it is expected to reduce the thermal efficiency and power output of the engine. Selected tests close in temperatures on the hot side of the engine using

Ø8 mm pellets are shown in Table 4 before and after 100 h of operation. The temperature ratio is increased slightly after 100 h of operation comparing the data for the Ø8 mm pellets for the first cycle of operation. This represents that the amount of heat supply to the engine is decreased, which is due to that the engine has reached a stabilized state. The effect of fouling is seen as a result of the cycling operation of the burner for the case 1 and 2 where there is an increase of the temperature ratio after each cycle. A decrease of the temperature ratio along the increase of the maximum temperature (T_2) is observed in Fig. 4a where no considerable effect of the pressure on the temperature ratio is seen. However, there is an effect of the pressure on the maximum temperature T_2 . This confirms the similarity between the temperature ratios shown in case 3 (Table 4) for the different pressures tested. Along the decrease of the temperature ratio, shown in Fig. 4a, higher thermal efficiency and power output are expected for the given pressures.

Effectiveness factors close to 1 indicates high thermal efficiency [36]. The effectiveness factor of the regenerator is somewhat higher for the Ø8 mm pellets than for the Ø6 mm wood pellets considering case 1 and 2 in Table 4. This indicates higher thermal efficiency for the Ø8 mm pellets. The effectiveness slightly decreases due to the cycling operation of the burner but there is not a clear relation with T_2 . Previous studies have only showed the relation of the effectiveness and thermal efficiency [13,36]. Fig. 4b shows the relation of the regenerator temperature gradient ($T_{11} - T_{12}$) and the effectiveness of the regenerator at different pressures after 100 h of operation. Here it is seen that the temperature ratio has no great impact on the effectiveness. The effectiveness is however raised at higher pressures. An increase in the temperature gradient has been reported to increase the performance of the Stirling engine due to an increase in the exchanged energy between the matrix and the working fluid of the regenerator [13]. Therefore, the power output is expected to be increased with higher temperature gradients at a given pressure for the cases shown in Fig. 4b. As a result of this relation, it is suggested that the effectiveness has no effect on the power output of the engine which confirms the statement of a previous study on another type of Stirling engine (alpha) [12]. Since maximum values of the effectiveness of the regenerator reached 0.55, a reduction of the porosity of the regenerator matrix could raise the performance of the engine due to a decrease in external conduction losses and an increase of the exchanged energy between the gas and the regenerator [13].

Table 4Temperature ratio (R), effectiveness of the regenerator (E_f) for Ø6 and Ø8 mm wood pellets before 100 h and for Ø8 mm wood pellets after 100 h at different operating pressures of the engine.

Case	Fuel type	Cycle	Press. [bar]	T_1 [°C]	T_2 [°C]	R	E_f
1. Before 100 h of operation	6 mm W	1 cycle	12	1013 ± 44	511 ± 21	0.39 ± 0.01	0.38 ± 0.003
		2 cycle		953 ± 38	462 ± 15	0.41 ± 0.01	0.36 ± 0.01
		3 cycle		937 ± 38	428 ± 19	0.43 ± 0.01	0.34 ± 0.01
2. Before 100 h of operation	8 mm W	1 cycle	12	1037 ± 21	573 ± 18	0.36 ± 0.01	0.41 ± 0.01
		2 cycle		1035 ± 19	555 ± 5	0.37 ± 0.02	0.40 ± 0.006
		3 cycle		1036 ± 18	542 ± 9	0.37 ± 0.04	0.39 ± 0.004
3. After 100 h of operation	8 mm W	1 cycle	12	1095 ± 39	578 ± 8	0.38 ± 0.01	0.48 ± 0.01
		1 cycle	15	1069 ± 46	574 ± 19	0.38 ± 0.002	0.48 ± 0.003
		1 cycle	17	1063 ± 12	579 ± 12	0.39 ± 0.01	0.53 ± 0.009

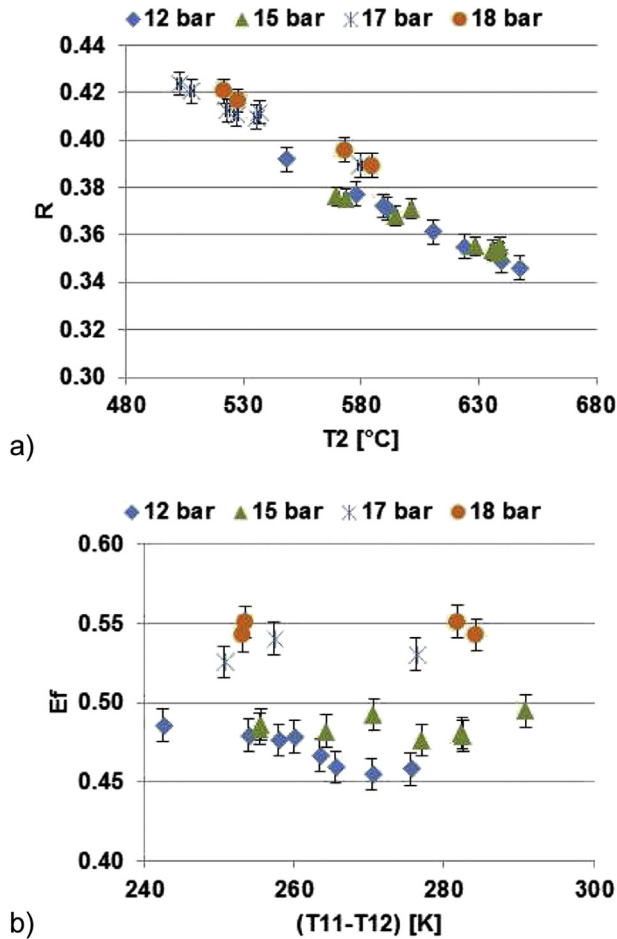


Fig. 4. a) Temperature ratio (R) as a function of the temperature on the hot side of the Stirling engine; and b) effectiveness of the regenerator (E_f) as a function of the regenerator temperature gradient at different pressures after 100 h of operation using 08 mm pellets (10–20 min test per temperature with a maximum standard deviation of 2%).

The overall thermal power absorbed by the Stirling engine as a function of the average temperature on the hot side and pressure is shown in Fig. 5. Higher temperature leads to higher thermal power absorbed by the engine at the same pressure according to Fig. 5. There is a positive relation between the hot side temperature and the thermal power for all the pressures tested. This positive relation between the temperature and thermal power can be associated to the linearity between electrical output and hot side and hot source temperatures found using beta [10] and gamma [11] engines. On the other side, the influence of the pressure on the thermal power is not clear since there is an overlap of thermal power measured between for example from 12 to 15 bar. However, it is possible to distinguish that the power output of the engine was higher between 12 and 15 bar in comparison to the measured values between 17 and 18 bar in the same range of temperatures (520–580 °C). Other experimental studies performed by Karabulut et al. [10] and Cinar et al. [11] have shown that the pressure follows a proportional tendency with the power output up to some point when it is not longer positively influencing the electricity production.

The scatter of data in Figs. 4 and 5 for a given pressure is due to that the pellet burner does not provide a completely fixed gas flow or fixed flame temperature. These fluctuate due to the complexity of the solid–gas phase combustion [37], which will affect the temperature of the hot side of the engine.

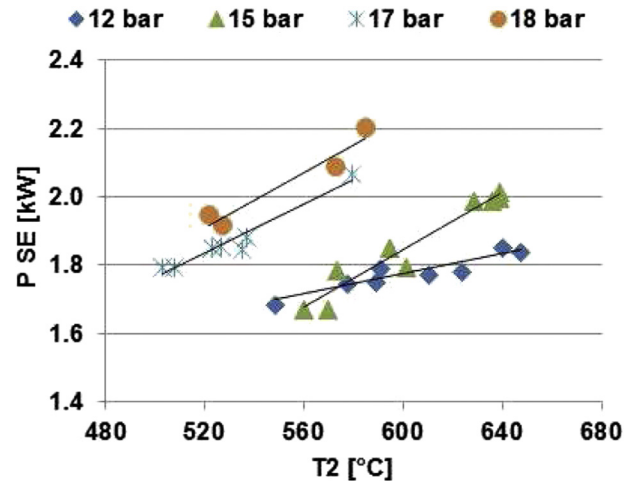


Fig. 5. Overall thermal power absorbed by the Stirling engine [kW] as a function of the hot side end temperature [T_2 °C] and the pressure [bar] for 08 mm wood pellets at duplicated tests (10–20 min test per temperature with a maximum standard deviation of 2%).

The experimental results presented in Figs. 3–5 can be used to validate numerical models to predict the performance of the Stirling engine coupled with an external heat source based on biomass.

3.3. Overall system evaluation under steady-state and transient conditions

The losses and power outputs in the system are shown in Fig. 6 under transient conditions for one cycle of operation of the burner at 12 bar engine pressure. Table 5 presents the relative losses to the fuel input power as well as the efficiency of the system integrated with the Stirling engine in comparison to previous results for the same residential heating boiler as a stand-alone unit [29]. Here a complete run cycle of three hours without interruption has been considered for 12 and 15 bar of the engine pressure, respectively.

There is a delay of the heat transfer from the hot flue gas to the water boiler and to the Stirling engine as shown in Fig. 6. This latter is due to the hot side temperature at which the engine starts working: 350 °C. The losses due to release of CO (chemical losses) are much lower than the thermal losses and can be negligible in evaluation of the overall system under transient conditions. The thermal losses are unavoidable due to the temperature of the gases

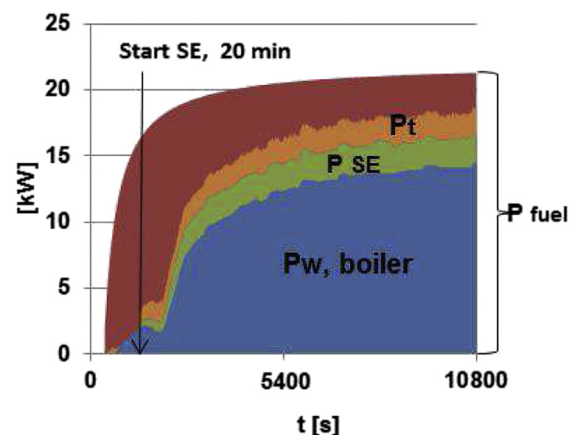


Fig. 6. Power input (P_{fuel}), power outputs (P_{SE} and $P_{w, boiler}$); and losses in the system (P_t) under transient conditions for 3 h test, 12 bar engine pressure.

Table 5

Comparison of the system with and without the Stirling engine integration (duplicated tests of 3 h).

	Residential heating boiler	Integrated system with SE	
<i>Type of fuel</i>	Ø8 mm wood pellets		
Stirling engine pressure [bar]	—	12	15
Thermal power [kW] (as hot water)	17 (±0.5)	15 (±1)	15 (±1)
Electrical power [W]	0	461 (±15)	489 (±23)
<i>Losses (as % of fuel input power)</i>			
P_{ch} [%]	0.03 (±0.02)	0.07 (±0.005)	0.05 (±0.02)
P_t [%]	7 (±0.5)	8 (±0.2)	9 (±0.2)
$P_{no\ identify\ losses}$ [%]	3 (±1)	20 (±5)	18 (±5)
<i>Emissions</i>			
NO _x mg/Nm ³ at 13% O ₂	54 (±8)	67 (±3)	61 (±9)
CO mg/Nm ³ at 13% O ₂	56 (±30)	99 (±30)	68 (±33)
O ₂ %	6 (±1)	8 (±0.4)	10 (±1)
<i>Efficiency</i>			
η_{system} [%]	89 (±2)	71 (±6)	72 (±6)

in the chimney should be kept above 100 °C preventing water condensation which may cause corrosion [28]. The gap between the fuel power input and quantified losses together with the power outputs represents non-identified losses which could be due to losses in the combustion chamber, in the connection pipe between the mentioned chamber and boiler and also losses inside the Stirling engine. The transient power outputs showed in Fig. 6 reveal the importance of the start-up of the burner and the engine since the engine starts around 20 min after the burner starts. The results in Fig. 6 are also of importance to validate models of annual heat and power production considering the mentioned delay in the start of the engine and if adequate fouling factors are found to simulate the cycling operation of the system. As seen in Section 3.1, the temperatures in the hot side of the Stirling engine are reduced due to the cycling operation and ash accumulation. Therefore, the thermal power absorbed by the engine is expected to decrease after the first cycle. A future study will include the magnitude of the fouling factor as result of the cycling operation and ash left onto the surface of the Stirling heat exchanger.

The results shown in Table 5 reveal an overall decrease of the system efficiency under steady-state when the Stirling engine is integrated. This is caused mainly by the nominated non-identified losses. The non-identified losses can be reduced by the installation of the burner inside the boiler keeping a closer distance between the burner and the engine heat exchanger. Slightly higher chimney losses are found when integrating the Stirling engine due to the limiting factor of the district cooling temperature and flow. The CO emissions are somewhat increased when integrating the engine to the system. This is suggested to be due to higher levels of oxygen in the flue gas (Table 5). These higher levels of oxygen decrease the temperature on the glow bed and therefore, the conversion rate of CO to CO₂ is also diminished [27]. The overall efficiency of the system measured in this study is similar to the data presented by Thiers et al. [5]. They found and overall system efficiency of around 72% for a micro-CHP plant based on a 3 kW_e alpha Stirling engine and fueled on wood pellets. Comparing that efficiency with the measured within this study, the type of Stirling engine has very little impact on the overall CHP efficiency and it is acceptable.

4. Conclusions

This study analyzed the influence of the type of pellets, the distance between the burner and Stirling engine hot side, and the cycling operation on the engine hot side temperature. The Stirling engine heat transfer efficiency was assessed according to the

temperature ratio and the effectiveness of the regenerator for different cases. The influence of the hot side engine temperature and pressure on the overall thermal power absorbed by the engine was also observed. Losses and thermal power outputs of the micro-CHP system were presented under transient conditions and the overall efficiency of the system was evaluated and compared with previous results for the residential boiler only [29].

The main conclusions drawn from this study are:

- The temperature in the hot side of the Stirling engine is very sensitive to the distance to the pellet burner and therefore this should be considered as a design condition. At closer distance, the radiative heat transfer plays an important role, while at further distance increase, convective heat transfer plays a larger role.
- Ash accumulation on the burner grate and over the surface of the Stirling engine heat exchanger causes a decrease of the hot side temperature after several cycles of pellet burner operation. Removal of ash after a certain amount of cycles is advisable.
- The temperature ratio of the Stirling engine is lower and the effectiveness of the regenerator is higher when using Ø8 mm pellets in comparison to Ø6 mm wood pellets. These shows that a higher amount of heat was supplied to the engine which in turns means as well a higher thermal efficiency when using Ø8 mm wood pellets.
- The temperature ratio of the Stirling engine is not affected by the pressure. The effectiveness of the regenerator is increased at higher pressures but it is not affected by the increase of the temperature gradient in the regenerator.
- The thermal power absorbed by the Stirling engine increases with the temperature on the hot side. In the same range of temperatures, higher pressures lead also to higher thermal power absorbed by the engine.
- The overall efficiency of the system is maintained above 72% when integrating the Stirling engine with a residential boiler producing both heat and power.

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Abbreviations and nomenclature

C	carbon content of the test fuel (wet basis) and in residue ash (% of mass)
Ca	calculated
CHP	combined heat and power
Cp	specific heat (kJ/kg K)
D	distance between the burner and Stirling engine hot side (cm)
d.b.	dry basis
Ef	effectiveness of the regenerator
H	hydrogen content of the test fuel (wet basis) (% of mass)
HHV	high heating value (MJ/kg)

LHV	low heating value (MJ/kg)
LPG	liquefied petroleum gas
$\dot{m}$	mass flow (kg/s)
M	moisture content in the fuel (wet basis) (% of mass)
P	power (kW)
R	temperature ratio of Stirling engine
SE	Stirling engine
T	temperature ($^{\circ}\text{C}$, K)
t	time (s)
vv	volume basis
W	wood
w.b.	wet basis
wt.	weight basis

Symbols

$\emptyset$	diameter (mm)
η	efficiency (%)

Subscripts

1	flame
2	hot side of the SE
3	flue gas at the exit of the box
4	flue gas before water heat exchanger
5	flue gas at the chimney
6	water inlet boiler
7	water outlet boiler
8	water inlet SE
9	water outlet SE
10	hot side of regenerator of the SE
11	cold side of regenerator of the SE
12	cold side of the SE
ch	chemical losses
e	electrical
FG	flue gas
h	hot side
md	dry flue gas in standard conditions
mH_2O	water vapor in standard conditions
s1	subsystem 1
s2	subsystem 2
s3	subsystem 3
t	thermal losses
w	water

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